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Test Results**

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ABSTRACT

Finger seals have significantly lower leakage rates than conventional labyrinth seals used in gas turbine engines and are expected to decrease specific fuel consumption by over 1 percent and to decrease direct operating cost by over 0.5 percent. Their compliant design accommodates shaft growth and motion due to thermal and dynamic loads with minimal wear. The cost to fabricate these finger seals is estimated to be about half the cost to fabricate brush seals. A finger seal has been tested in NASA's High-Temperature, High-Speed Turbine Seal Test Rig at operating conditions up to 1200 °F, 1200 fps and 75 psid. Static, performance, and endurance test results are presented. While seal leakage and wear performance are acceptable, further design improvements are needed to reduce the seal power loss.

INTRODUCTION

A variety of seals are used by the gas turbine industry to contain and direct secondary flow into and around components for cooling, and to limit leakage into and from bearing and disc cavities. The function of these seals is very important to the component efficiencies and attendant engine performance.¹ The Joint Turbine Advanced Gas Generator-phase 3 (JTAGG III) program goals are to reduce overall engine specific fuel consumption by 40 percent, increase engine shaft horsepower to weight ratio by 120 percent, reduce production cost by 35 percent, and reduce maintenance cost by 35 percent.² Improved seals will be needed to reach these goals.

The finger seal is an innovative design recently patented³ by AlliedSignal Engines, which has demonstrated considerably lower leakage than commonly used labyrinth

seals and is considerably cheaper than brush seals. The cost to produce finger seals is estimated to be about half of the cost to produce brush seals.³ Replacing labyrinth seals with finger seals at locations that have high pressure drops, typically main engine and thrust balance seals, can reduce air leakage at each location by 50 percent or more. This directly results in a 0.7 to 1.4 percent reduction in specific fuel consumption and a 0.35 to 0.7 percent reduction in direct operating cost.⁴

In the late 1990s a low cost, pressure-balanced, low hysteresis finger seal was developed and successfully demonstrated at operating conditions of 778 fps, 60 psid, and 1000 °F and 945 fps, 80 psid, and 800 °F. Both the seal and rotor were in excellent condition after 120 hours of endurance testing.⁴ The finger seal is a contacting seal, which raises concern about the heat it will generate and its life capability at the higher temperatures and speeds required for advanced engines. To address this concern a pressure-balanced, low hysteresis finger seal was tested at operating conditions up to 1200 fps, 75 psid and 1200 °F. These are the first test results obtained with NASA's new High-Temperature, High-Speed Turbine Seal Test Rig. The test hardware, apparatus, and experimental procedures will be described followed by a discussion of the seal performance and wear results.

TEST HARDWARE

Using design criteria developed during earlier testing of various finger seal configurations, a low hysteresis finger seal was designed and fabricated for testing at advanced engine operating conditions. The low hysteresis, pressure-balanced seal design features developed in reference 4 were incorporated in this design.

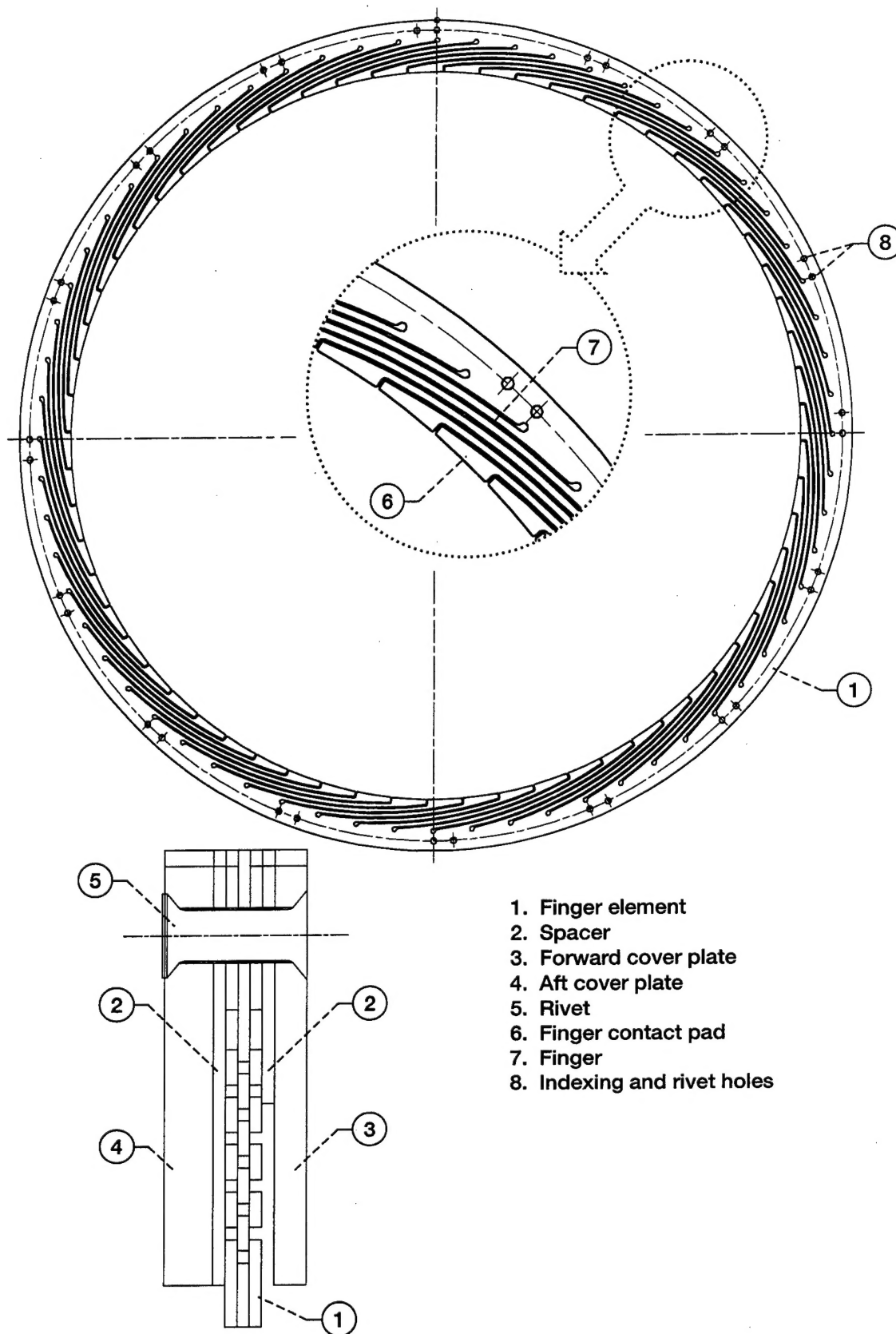


Figure 1.—Finger seal design.

The finger seal is similar in general configuration to a brush seal, but functions in a different manner. Instead of a random array of fine wires, the finger seal uses a stack of tight-tolerance sheet stock elements. Each element is machined to create a series of slender curved beams or *fingers* around the inner diameter (fig. 1). Each of these fingers (7) has an elongated contact pad (6) at its free end. Each element (1) also has a series of assembly hole pairs (8) near its outer diameter. These holes are for the rivets (5) that assemble the seal. The holes are spaced such that when the elements are alternately indexed to the two holes, the spaces between the fingers of one element are covered by the fingers of the adjacent element. Usually a seal is assembled with multiple finger elements (1), forward and aft spacers (2) and forward (3) and aft (4) cover plates. The seal is fitted over the rotating shaft or rotor with a small amount of clearance or interference, depending on the application. The staggered finger/pad features as well as the radial contact between the rotating land and the pads impede airflow through the seal. The flexible fingers can bend radially to accommodate shaft excursions and relative growth of the seal and rotor resulting from rotational forces and thermal mismatch.

A key feature of the finger seal is its low cost of manufacture. The geometric features on the seal laminates are fashioned using wire electric discharge machining, which is extremely cost-effective. Sheet stock of various alloys and thickness required for the seals is readily available. Riveting of the assembly does not require any elaborate tooling or assembly process.

A parametric finite element modeling program and Honeywell proprietary programs were used to optimize the finger seal design. The finger seal tested was sized to run with a slight interference at operating conditions. The finger elements, spacers and side plates are made of sheet AMS5537. This is a cobalt-base alloy that combines good formability and excellent high temperature properties. It displays excellent resistance to the hot corrosive atmospheres encountered in jet engine operations.

The 8.5-in. diameter test rotor is made of MAR-M-247, a nickel-base alloy with excellent high temperature properties. The seal runner surface on the rotor is coated with chrome carbide using high velocity oxygen fuel thermal spraying.

TEST APPARATUS

Turbine Seal Test Rig

Testing was conducted in the NASA High Temperature, High Speed Turbine Seal Test Rig shown in figure 2 and

located at the Glenn Research Center in Cleveland, Ohio. The turbine seal test rig consists of an 8.5-in. diameter test rotor mounted on a shaft in an overhung configuration. The shaft is supported by two oil-lubricated bearings. A balance piston controls the axial thrust load on the bearings due to pressure loads on the test rotor. An air turbine drives the test rig. A torquemeter is located between the air turbine and the test rig and is connected to each by quill shafts. The test seal is clamped into the MAR-M-247 seal holder as shown in figure 3. A C-seal located at the seal holder/test seal interface prevents flow from bypassing the test seal at its outer diameter. The seal holder is heated to approximately match the thermal growth of the rotor and prevent a damaging change in radial clearance. Heated, filtered air enters the bottom of the test rig and passes through an inlet plenum that directs the heated air axially toward the seal-rotor interface. The hot air either leaks through the test seal to the seal exhaust line or exits the rig before the test seal through a controlled bypass line at the top of the rig. If seal leakage is low, the bypass line must be open to maintain sufficient flow through the test rig to keep the rig hot.

Instrumentation

Seal inlet and exit temperatures and static pressures, seal upstream metal temperature, and seal backface temperatures were measured at the locations shown in figure 3. For each measurement there were 3 probes equally spaced around the circumference, except for the upstream seal metal temperature for which 2 thermocouples were located at the 90 and 180° positions (0° is top dead center). Type-K thermocouples were used and all were 0.062 in., Inconel sheath, closed ball except the seal exit temperatures, which were 1/8-in. diameter and the seal metal and backface temperatures, which were open-ball.

High temperature, capacitance proximity probes were mounted in the seal holder at four, equally spaced locations to view the test rotor outer diameter. These probes were used to measure the change in clearance between the seal holder and the rotor and to monitor the rotordynamic behavior of the test rotor. The average inlet air temperature is used as the probe temperature when correcting the probe output. These proximity probes have an accuracy of 0.0002 in. at room temperature.

Annubar flow meters are used to measure the flow rates of the hot air supplied to the rig and the air exiting the rig through the bypass line. The seal leakage is the difference between these two flow measurements. The seal leakage rate is then used to calculate the flow factor, which is defined as:

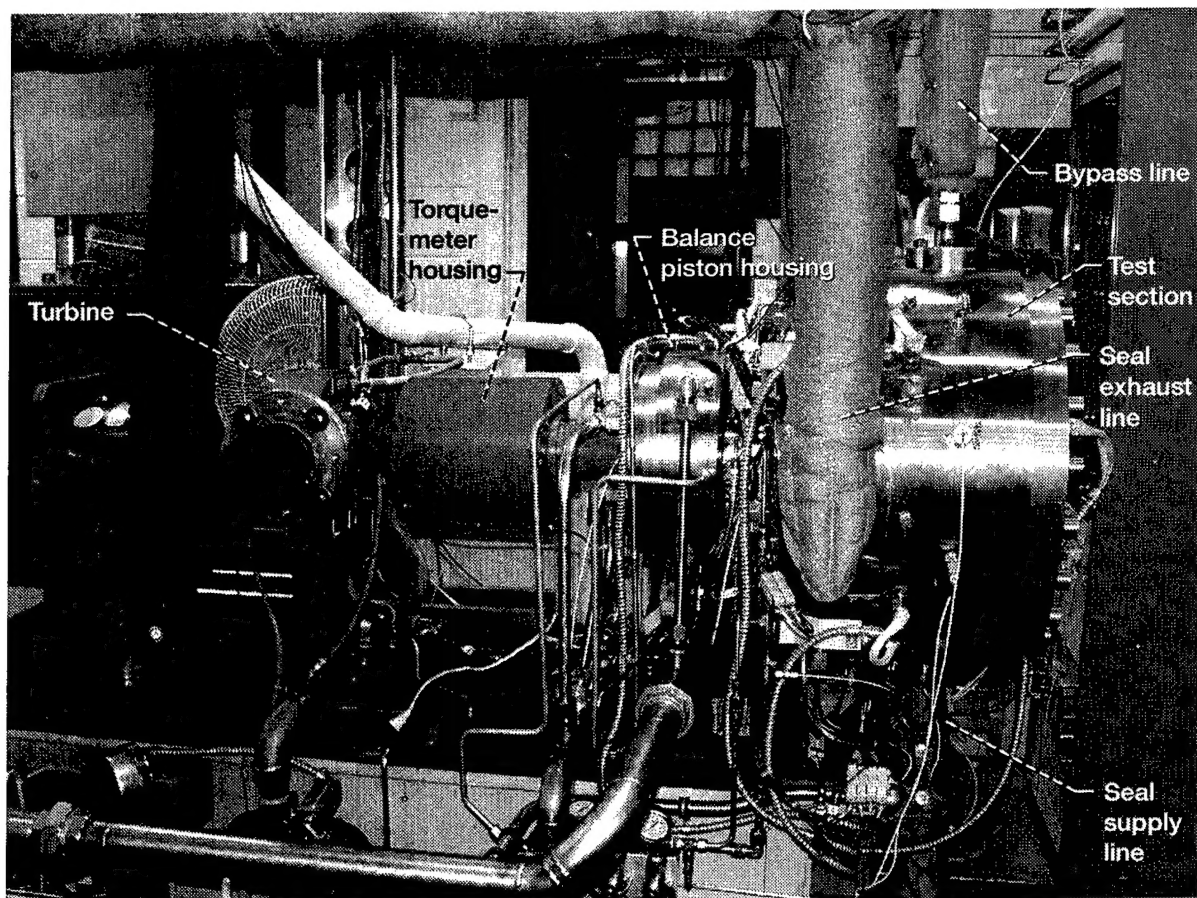


Figure 2.—High-temperature, high-speed turbine seal rig.

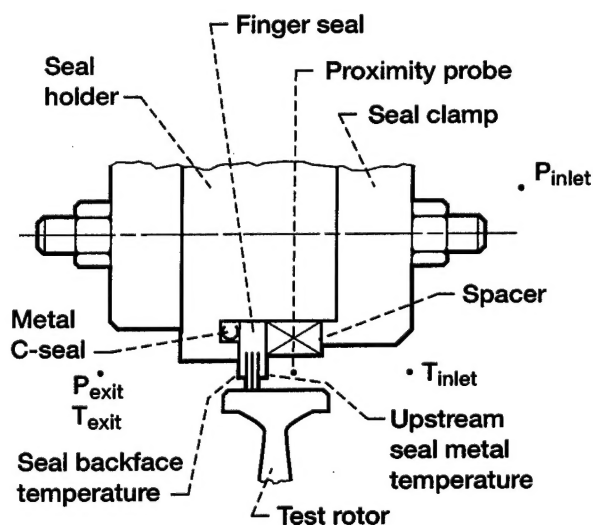


Figure 3.—Test seal configuration and location of research measurements.

$$\Phi = \frac{\dot{m} \sqrt{T_{avg} + 459.60}}{P_u \times D_{seal}}$$

where

\dot{m} Air leakage flow rate, lbm/sec
 T_{avg} Average seal air inlet temperature, °F
 P_u Air pressure upstream of seal, psia
 D_{seal} Outside diameter of the seal rotor, in.

The flow factor can be used to compare the leakage performance of seals with different diameters and with different operating conditions. The accuracy of the measured flow factor is ± 1.5 percent.

A phase shift torque meter measures the total torque of the seal test rig. It has a feature to compensate for any relative motion between the torsion shaft and stator. The torque meter is rated to 16 ft-lb, has a maximum operating speed of 50,000 rpm, and an absolute accuracy of 0.13 percent or 0.021 ft-lb. The seal torque is the rig torque

with the test seal installed minus the rig tare torque. The rig tare torque was measured at various inlet air temperatures and speeds with no seal installed. This data was two-dimensionally curve fitted. The fitted curve is used with the measured average inlet air temperature and speed to infer the corresponding tare torque. Seal power loss is simply the seal torque multiplied by speed. The maximum error in the seal power loss measurements is 0.131 hp over the range of test conditions. The speed measurement from the torque meter is accurate to <0.04 percent or 13 rpm at the maximum speed tested.

EXPERIMENTAL PROCEDURES

Four tests were performed: a static leakage test, a performance test, an endurance test, and a post-endurance performance test.

A static test was performed at ambient temperature, 800 °F, and 1200 °F to obtain baseline leakage data. At steady conditions, seal leakage data was taken for 1 minute at steady seal differential pressures from 0 to 75 psid and then from 75 to 0 psid.

Seal performance test data were taken at average inlet air temperatures of 800, 1100, and 1200 °F. At each temperature, differential pressures of 10, 40, and 75 psid were applied and at each pressure, surface speed was stepped up and down as follows: 0, 600, 900, 1200, 900, 600, and 0 fps. Data were recorded every second for approximately 30 seconds after reaching a steady state at all test conditions, except for the first test condition of 800 °F and 10 psid for which data was continuously recorded every second to observe the initial wearing of the seal. The seal and rotor were inspected after the performance test was completed.

The endurance test assessed the ability of the seal to maintain low leakage over an extended period of time. This test was conducted at 1200 °F, 1200 fps, and 75 psid for four hours at which time the seal leakage and power loss stopped changing. The seal and test rotor were removed for inspection after 1, 2, and 4 hours of testing. The performance test was repeated after the endurance test.

SEAL PERFORMANCE RESULTS

Initial Static Test

The static performance of the finger seal at an average inlet air temperature of 1200 °F is shown in figure 4 as a plot of flow factor versus pressure drop across the seal. The flow factor increases with pressure drop until about 15 psid and then levels off at a flow factor of approximately 0.0015. At this point the flow is choked.

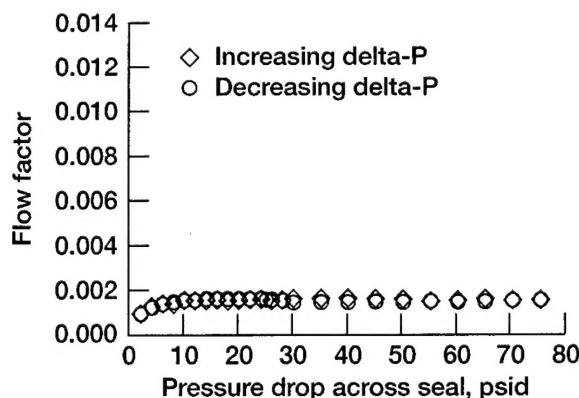


Figure 4.—Static leakage performance of finger seal at average seal inlet air temperature of 1200 °F.

Performance Test

Initial Rotation. The finger seal was tested at 800 °F average inlet air temperature and 10 psid across the seal. Data were recorded once a second while the speed was stepped up to obtain 600, 900, and 1200 fps surface velocities and then stepped back down to 900, 600, and 0 fps. Figure 5 shows a time history plot of shaft speed, seal power loss, average seal backface and inlet air temperatures, and radial seal clearance for this initial rotation. The seal power loss shown is the total power minus the steady-state tare power. The positive and negative spikes as each speed level is attained are largely due to acceleration and deceleration inertia. Note that at 1200 fps the steady state seal power loss declines as time at condition increases. This could indicate wearing of the seal. At steady-state conditions the seal power loss did not exceed 2.8 hp. Frictional heating due to seal-to-rotor contact and wearing is also evident in the average seal backface and inlet air temperatures time history. As speed is decreased, both the seal power loss and average seal backface temperature levels are lower than during the speed increase, which might indicate that wear has occurred. The centrifugal growth of the rotor can be seen in the time history of the radial seal clearance. The radial clearance shown in figure 5 is the change in the distance between the seal holder and the rotor from ambient, static conditions, plus the initial clearance between the seal and the rotor at ambient temperature. This clearance is not necessarily the clearance between the fingers and the rotor. From 0 to 1200 fps the radial interference increased from 0.0032 to 0.0098 in., a change of 0.0066 in., which exceeds the expected centrifugal growth of about 0.0033 in. This indicates that the seal holder and rotor are not maintaining the same temperature, which is supported by the approximately 200 °F change in the seal backface temperature during this initial rotation.

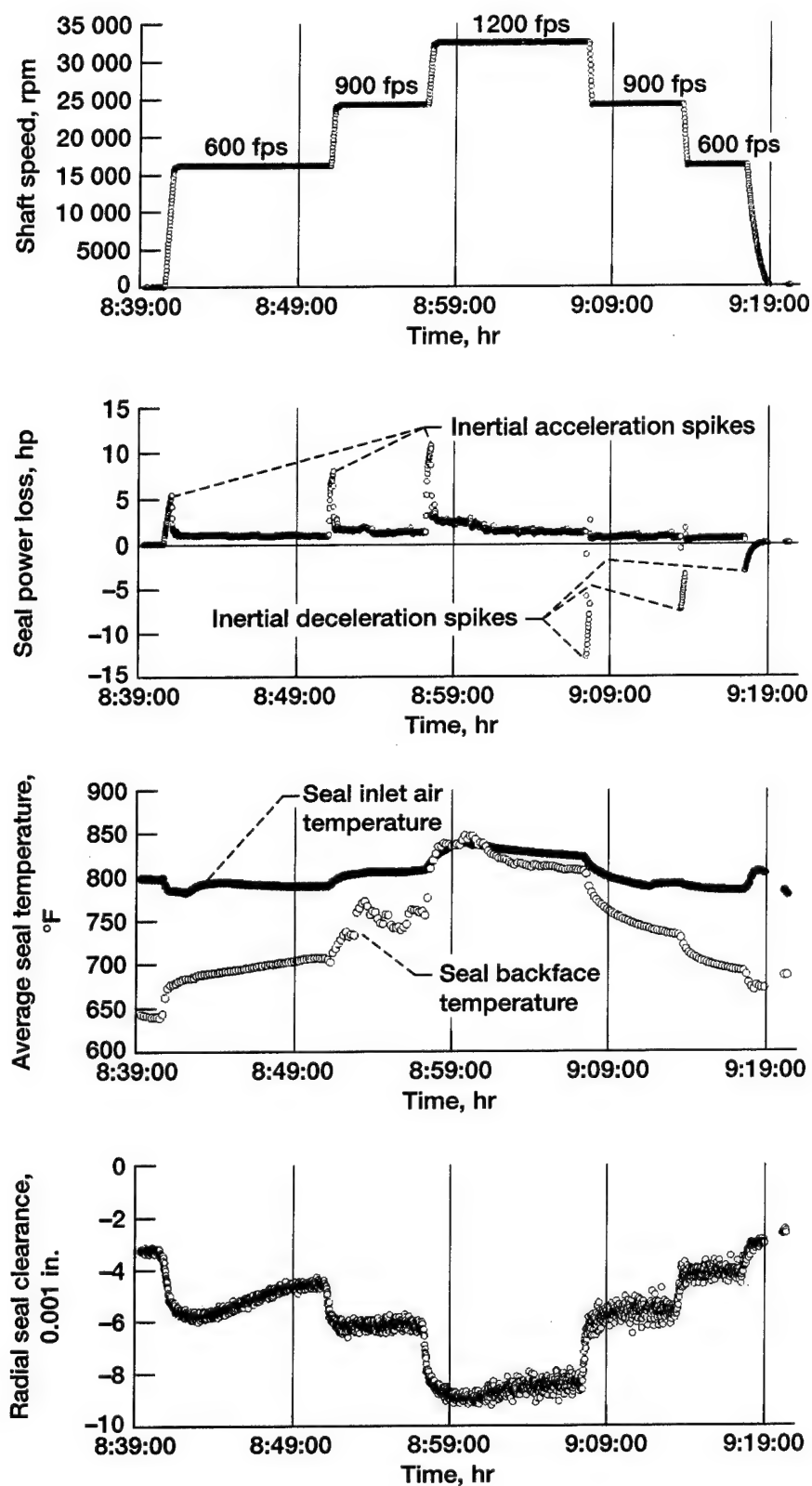


Figure 5.—Time history of initial rotation at 800 °F average seal inlet air temperature and 10 psid pressure drop across seal.

Leakage. The finger seal leakage performance at average seal inlet air temperatures of 800 and 1200 °F is shown in figures 6 and 7, respectively, as plots of flow factor versus surface velocity at pressure drops across the seal of 10, 40, and 75 psid. At 800 °F it can be seen that the flow factor at 10 psid is less than at 40 and 75 psid. Also, the flow factor data at 10 psid and for increasing speed at 40 and 75 psid are about the same, approximately 0.0016 to 0.0018. Hysteresis can be seen in the data taken at 40 psid and at 75 psid and is more pronounced at 75 psid. When speed increases, the centrifugal growth of the rotor pushes the fingers radially away. When speed decreases and the rotor diameter shrinks, the fingers may remain in their outer position causing leakage and flow factor to increase. Hysteresis may also be due to rapid wear of the seal during initial shaft rotation, which would increase the seal clearance. Likewise, seal holder and rotor temperature

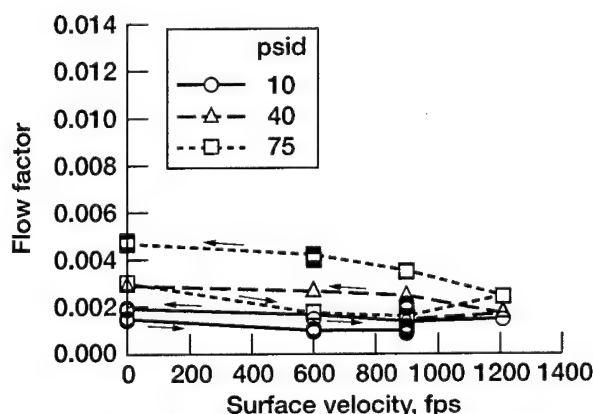


Figure 6.—Finger seal performance test data at 800 °F average seal inlet air temperature and 10, 40, and 75 psid pressure drop across seal.

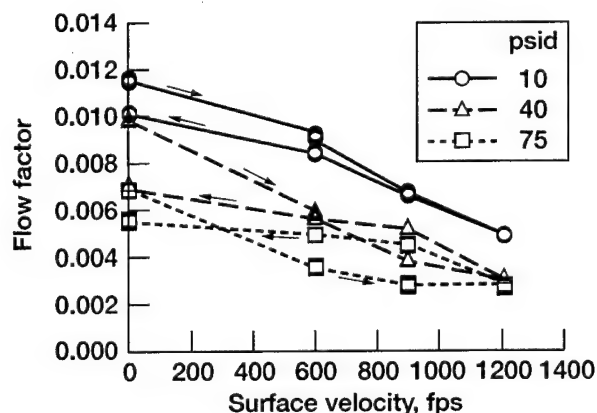


Figure 7.—Finger seal performance test data at 1200 °F average seal inlet air temperature and 10, 40, and 75 psid pressure drop across seal.

changes can affect the seal clearance and appear as hysteresis. However, even with the hysteresis the flow factor is still at an acceptable level of less than 0.006.

The flow factor for an average inlet air temperature of 1200 °F is shown in figure 7. At all three pressure differentials, the flow factor decreases as speed increases due to the centrifugal growth of the rotor reducing the clearance, as expected. Again, hysteresis is evident, but at 10 and 40 psid there is an inconsistency with previous data in that flow factor is lower for decreasing speed than for increasing speed. Also, as pressure drop across the seal increased, flow factor decreased, which is opposite to what happened at 800 °F. This can be explained by a look at the corresponding radial clearance data in figure 8. It shows that the radial clearance is lower for decreasing speed compared to increasing speed and that the radial clearance decreases as pressure drop across the seal increases. In this test sequence there is a clear and definite influence of changes in the radial clearance between the seal holder and the seal rotor. While data was taken at steady state conditions, the clearance between the seal holder and the rotor was not a controlled parameter. It seems that some of the hysteresis observed may actually be due to changes in the radial clearance between the seal holder and rotor and not due to the fingers getting stuck in the open position. The clearance data shown here indicates that a radial interference existed between the seal and rotor for most of the performance testing. By the end of the performance test the seal had worn such that a clearance existed between it and the rotor at ambient conditions.

Power Loss. The finger seal power loss is shown in figure 9 as a function of speed for the performance data taken at 800 and 1200 °F average inlet air temperature and

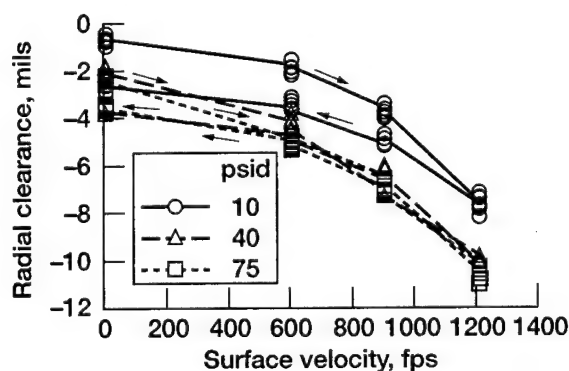


Figure 8.—Finger seal performance test radial clearance data at 1200 °F average seal inlet air temperature and 10, 40, and 75 psid pressure drop across seal.

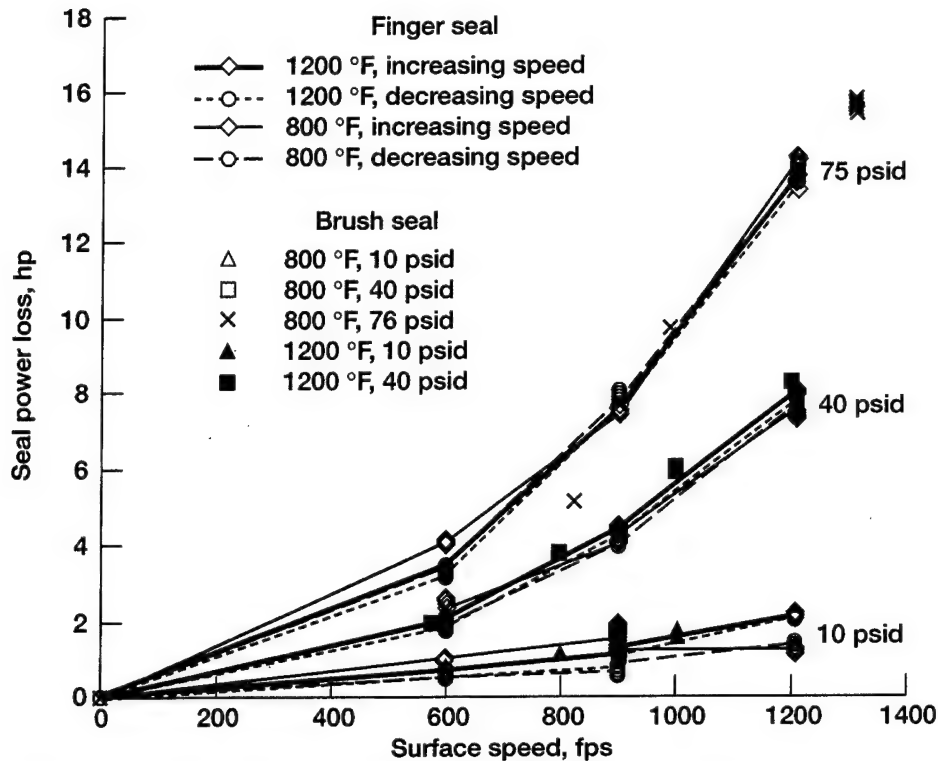


Figure 9.—Finger seal and brush seal power loss versus speed at 800 and 1200 °F average seal inlet air temperature and 10, 40, and 75 psid pressure drop across seal.

10, 40, and 75 psid. There is no significant difference between the data for 800 and 1200 °F. As expected, the seal power loss increased with speed and also increased as pressure drop across the seal increased due to pressure loading. Pressure loading occurs because the outer surface of the finger is longer than the inner surface, which under uniform pressure results in a net force pushing the fingers in towards the rotor. While the upstream finger element experiences uniform pressure loading, the pressure loading on the middle and downstream finger elements is not precisely known. The finger seal power loss at 1200 fps was 2, 8, and 14 hp at 10, 40, and 75 psid across the seal, respectively. The measured power loss was in good agreement with analytical predictions made by Honeywell using a proprietary code. A brush seal with a similar radial interference as this finger seal was also tested and its measured power loss is also shown in figure 9. The brush seal power loss is very similar to the finger seal power loss.

Endurance Test

The time history of key parameters for the endurance test is shown in figure 10. Surface speed and pressure drop across the seal are very stable over the run: 1200 ± 0.5 fps and 75 ± 0.09 psid. There is some small fluctuation in the flow factor, average inlet air temperature, and radial

clearance, and they correlate with each other. Seal power loss initially climbs and then levels out. The minimal change in flow factor over the duration of the mini-endurance test indicates that most of the wear of the seal occurred during the prior performance test. The final flow factor level of 0.004 is an acceptable leakage performance. It was also observed that the inlet air temperatures measured at three locations around the seal showed a more uniform temperature around the seal than during the performance test. In the endurance test the seal inlet temperature at the top of the rig was about 50 °F lower than at the 120 and 240° locations, compared to 100 to 150 °F for the performance test. This was largely due to the long period of time at constant conditions.

Figure 10 shows a flow factor of approximately 0.004 at 1200 °F, 75 psid, and 1200 fps after 4 hours of endurance testing. This is very similar to the low leakage results for a 5.1 in. diameter pressure-balanced finger seal design tested at 1000 °F, 60 psid, and 778 fps in reference 4.

Post-Performance Test

The performance test was repeated after the endurance test. Hysteresis was present in all the data taken at 800, 1100, and 1200 °F. Again, the flow factor decreased as

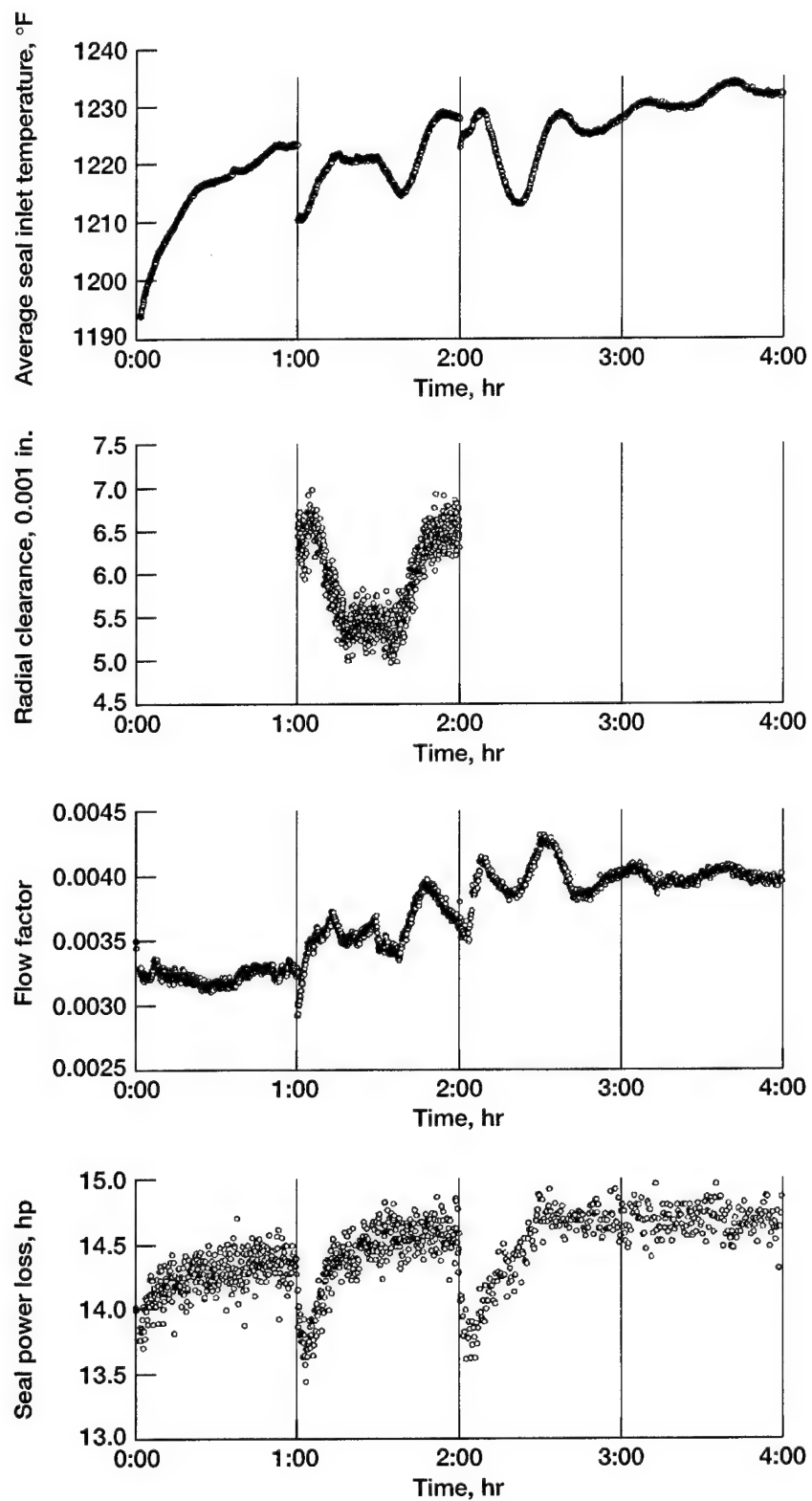


Figure 10.—Time history of finger seal endurance test at 1200 °F average seal inlet air temperature, 1200 fps surface speed, and 75 psid pressure drop across seal.

Table I.—Comparison of first and last performance test flow factors at 1200 fps

Avg. Inlet air Temp. (°F)	800			1100			1200		
Pressure Drop Across Seal (psid)	10	40	75	10	40	75	10	40	75
Flow factor First test	0.0015	0.0017	0.0025	0.0035	0.0026	0.0025	0.005	0.003	0.0029
Flow factor Last test	0.005	0.0042	0.0043	0.0068	0.0047	0.004	0.0082	0.0059	0.0047
Last ϕ /First ϕ	3.33	2.47	1.72	1.94	1.81	1.6	1.64	1.97	1.62

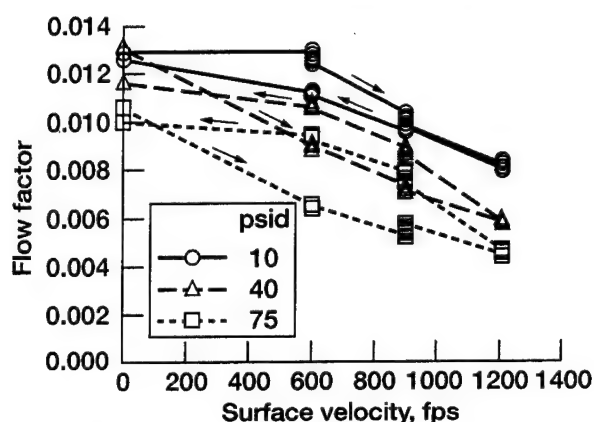


Figure 11.—Post-endurance performance test. Finger seal flow factor versus speed at 1200 °F average seal inlet air temperature and 10, 40, and 75 psid pressure drop across seal.

speed increased due to the centrifugal growth of the rotor and the pressure closing effect was evident in all cases. A comparison of the flow factors from the first and last performance tests at 1200 fps for all average inlet air temperatures and pressure drops across the seal is given in table I. The flow factors after the endurance test were 1.6 to 3.33 times those in the first performance test. The largest increase was between the initial data taken at 800 °F and 10 psid due to the wearing in of the seal.

The flow factors measured at an average inlet air temperature of 1200 °F are shown as a function of speed at pressure differentials of 10, 40, and 75 psid in figure 11. The flow factors ranged from 0.0045 to 0.013, which exceeds the goal of 0.006, but are still acceptable. The hysteresis is somewhat worse in this last performance test than in the first performance test. As seen in the first 1200 °F performance test, the hysteresis is reversed at 10 psid with the flow factor for decreasing speed being lower than for increasing speed. This is likely due to changes in the radial clearance, however the proximity probes stopped working during this test and the data is not available to confirm the effect.

The finger seal power loss measured during the post-endurance performance test is slightly less and very comparable to the measurements made in the first performance test as can be seen by comparing figure 12 to figure 9, respectively.

WEAR RESULTS

Seal Wear

The majority of the observed finger seal wear most likely occurred during the initial performance test, when the seal initial radial interference of 0.0065 in. changed due to centrifugal growth of the rotor and due to the pressure closing effect of the finger design.

Figure 13 is a plot of the accumulated seal weight loss after the first performance test (3.5 hr), 1st, 2nd, and 4th hour of endurance testing, and final performance test (11.0 hr). Over 70 percent of the total seal weight loss occurred during the first performance test. The remaining 30 percent was spread out in the remaining endurance and final performance tests. The seal weight loss appears to converge asymptotically toward a steady state value. Assuming that this weight loss occurred uniformly around the circumference of the seal, the radial wear is calculated to be about 0.035 in., or slightly more than half the finger pad thickness.

The visual inspection of the individual finger pads also suggests that minimal seal weight loss occurred after the first performance test. All 64 pads on each of the 3 laminates were given a qualitative rating (0 to 5 scale) based upon the amount of radial wear seen. For example, a '5' was given if the pad showed very little wear. A '3' was given for pads showing approximately 50 percent radial wear. A '1' was given if the pad toe was worn to a point. The 64 individual pad ratings were averaged to give an overall rating for each laminate. Figure 14 shows the averaged visual laminate wear rating for the 3 laminates of the finger seal after 1, 2, and 4 hours of endurance testing. The average laminate wear rate for each laminate decreases asymptotically to a steady-state value. Note that the middle

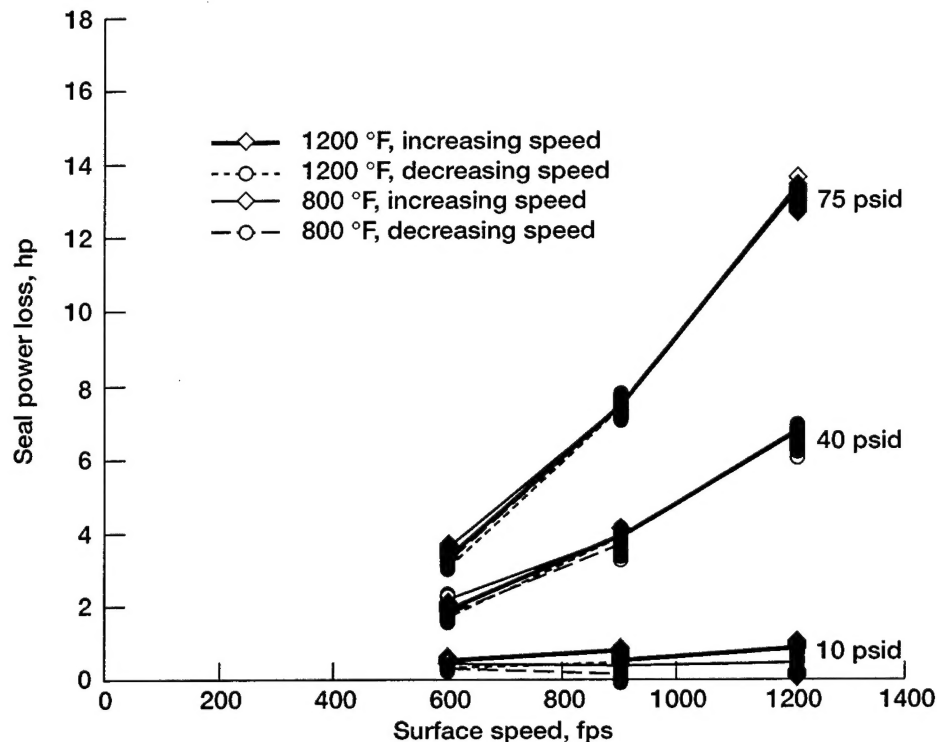


Figure 12.—Post-endurance performance test. Finger seal power loss versus speed at 1200 °F average seal inlet air temperature and 10, 40, and 75 psid pressure drop across seal.

Visual inspection rating definitions

- 5 = 0% average pad wear
- 4 = 25% average pad wear
- 3 = 50% average pad wear
- 2 = 75% average pad wear
- 1 = 100% average pad wear
- 0 = No pad

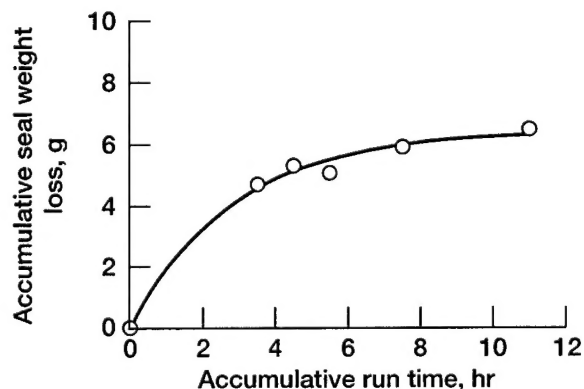


Figure 13.—Finger seal accumulative weight loss versus accumulative run time.

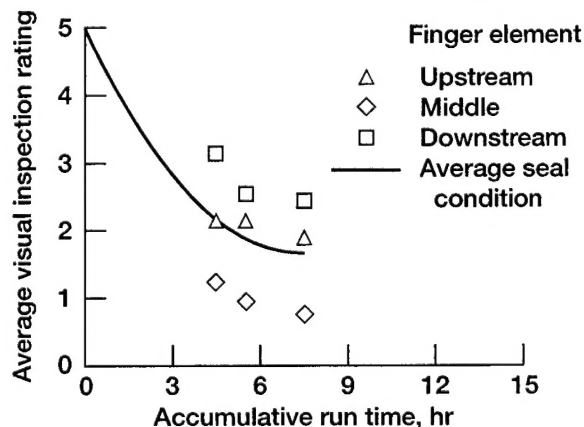


Figure 14.—Average visual inspection rating of finger seal wear versus accumulative run time for upstream, middle, and downstream finger elements.

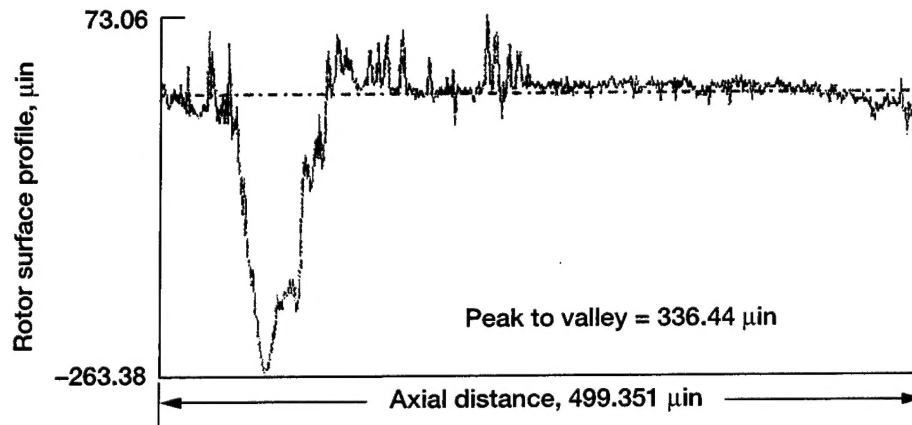


Figure 15.—Typical profile of seal wear track on rotor outer diameter.

Table II.—Average rotor wear track measurements

Test type	Width, in.	Depth, μin.
Baseline	0	—
First performance	0.095	245
First hr endurance	0.083	176
Second hr endurance	0.104	165
Fourth hr endurance	0.095	249
Second performance	0.096	216

laminate was observed to have the worst overall wear of the 3 laminates. The qualitative visual inspection ratings correspond reasonably well with the radial wear calculated from the weight loss.

Rotor Wear

Rotor wear was quantified using a profilometer. A typical profile of the wear track is shown in figure 15. Eight measurements were taken around the circumference of the rotor to determine an average track width and depth. These averages are presented in table II for each inspection. Both the track width and track depth measurements indicate that the majority of the seal wear took place during the first performance test. The average track width ranged from 0.080 to 0.100 in. and the average track depth ranged from 150 to 250 μin. This is a small and acceptable amount of

wear. The scatter in the data is likely due to the uncertainty in taking the measurements at the same circumferential location on the rotor for each inspection. The circumferential locations were visually sighted using the bolt hole locations and etch marks as guides. Given that the performance test effectively covers the entire range of temperatures, pressures, and surface speeds that the seal would be subjected to during the test program, it is likely that the overall seal track width was worn in during this first performance test.

CONCLUSIONS

1. The seal leakage performance is very sensitive to clearance.
2. The seal leakage performance is acceptable for advanced engines with the flow factor remaining <0.006 over most of the required operating conditions and maintaining a flow factor <0.004 during the endurance test, which simulated expected advanced engine rated power conditions.
3. The seal exhibited some hysteresis, some of which may actually be due to changes in the radial clearance between the seal holder and the test rotor.
4. The maximum finger seal power loss, which occurred at 1200 fps and 75 psid across the seal was 14 hp. Further design improvements can be made to reduce the seal power loss.
5. Finger seal power loss is comparable to the brush seal power loss.
6. Most of the seal wear occurred in the initial performance test. The rate of wear is acceptable. The HVOF Chrome carbide coating performed well, with a wear track depth less than 0.00025-in.

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